MULTIMEDIA UNIVERSITY

STUDENT ID NO

MULTIMEDIA UNIVERSITY

FINAL EXAMINATION

TRIMESTER 3, 2021/2022

EME3236 – HEATING, VENTILATION AND AIR CONDITIONING SYSTEMS

(ME)

15 AUGUST 2022
09.00 a.m. - 11.00 a.m.
(2 Hours)

INSTRUCTIONS TO STUDENTS

1. This question paper consists of six pages (including the cover page) with four questions and an Appendix.

2. Answer ALL four questions.

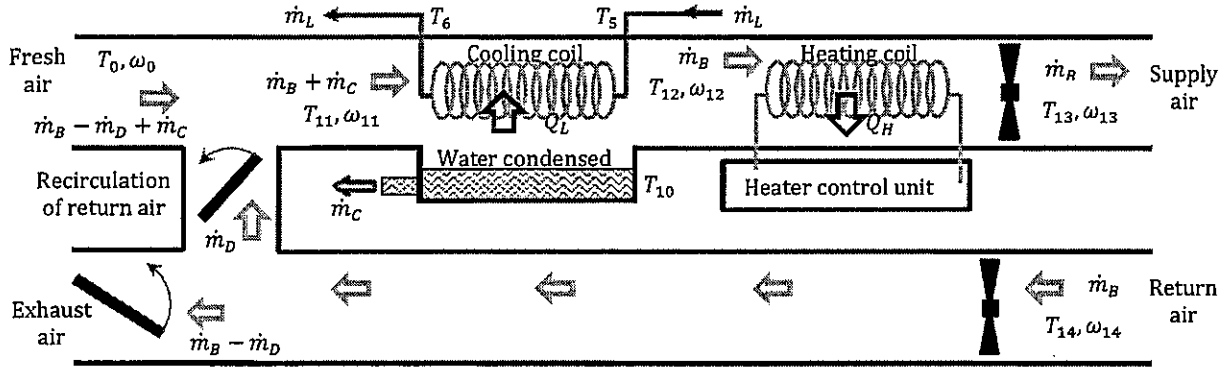
3. Each question carries 25 marks and the distribution of the marks for each question is given in brackets **[ ]**.

4. Write all your answers in the answer booklet provided.

5. Property-tables booklet is provided for your reference.

Question 1

Consider an adiabatic air handling unit (AHU), which has supply and return duct, connected by a recirculation section as shown in the following figure. The damper at the recirculation section is closed, where, $\dot{m}_D = 0$, so that the fresh air (State 0) at temperature 30 °C and absolute humidity 0.020 is directly supplied to the AHU. The fresh air is cooled by cooling coil to 15 °C (State 12), and then heated to human comfort condition at 20 °C (State 13). The return air (State 14) temperature and absolute humidity are 25 °C and 0.015 respectively. Moist air is supplied from the AHU at the rate of, $\dot{m}_D = 360$ liter/min. The chilled water enters the cooling coil at 10 °C and leaves at 15 °C, while the condensed liquid water exits the AHU at 15 °C. The flow throughout the AHU is maintained at atmospheric pressure, $p_0 = 100$ kPa. Assume the constant pressure specific heats of moist air, $c_{pm} = 1.015$ kJ/kg·K and liquid water, $c_{pw} = 4.2$ kJ/kg·K. The moist air density is 1.2 kg/m^3.



a. Calculate the rate of liquid water condensation.

[5 marks]

b. Determine the mass flow rate of chilled water through the cooling coil.

[4 marks]

c. Calculate the rate of heat rejection at cooling coil, the rate of heat addition at heating coil and cooling load of the AHU.

[6 marks]

d. Find the relative humidity of the supply air.

[2 marks]

e. If the damper at the recirculation section is fully open while the damper at the exhaust section is closed, the air now under internal circulation. Determine the new mass flow rate of the chilled water through the cooling coil in order to maintain the same temperature of states 12 and 13. Hence find the energy saving for recirculation of air in AHU. Also find the rate of water condensation. The effectiveness of the cooling coil is assumed to be equal to the case of no recirculation.

[8 marks]

Continued...

Question 2

In a particular section of a hotel, the layout of the floor plan with the information of air conditioning ducting network is shown in the following figure. The supply ducting of air conditioning is receiving cool air from AHU at 120 liter/s. At the diffuser of each room, volume flow rate of air supply is controlled by the damper so that it is evenly distributed to all the rooms, which is 40 liter/s. The damper at one of the diffuser is fully open so that there is no head loss on that diffuser.

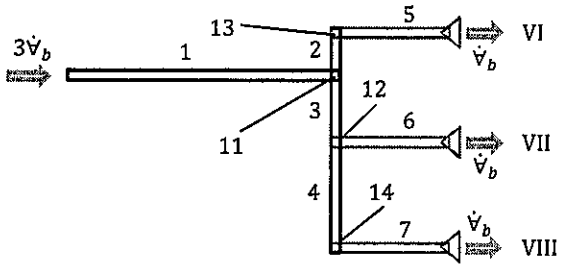


Figure Q2

Section	1	2	3	4	5	6	7
Diameter, D (m)	0.5	0.5	0.5	0.5	0.2	0.2	0.2
Length, L (m)	8.7	1.5	2.1	3.6	3.0	3.0	3.0

The local loss at tee and elbow are:
$K_{11b} = 1.00$; $K_{13} = 0.50$; $K_{12b} = 0.75$; $K_{12m} = 0.05$; $K_{14} = 0.50$; $K_{11m} = 1.00$
Gravitational acceleration is 9.81 m/s^2. The density and kinematic viscosity of air are 1.2 kg/m^3 and 1.5×10^{-5} m^2/s respectively.

a. Draw a schematic diagram of the ducting network.

[4 marks]

b. Find the air velocity at every section of the duct.

[5 marks]

c. Use Blasius formula to calculate the friction factor and determine the head loss at every section of the duct.

[11 marks]

d. Calculate the head loss at the diffusers.

[3 marks]

e. Determine the total head loss of the duct network.

[2 marks]

Continued...

Question 3

A side wall of the house is made of wood fiberboard, 5 m x 5 m x 0.02 m. The owner of the house decided to replace the wall with either using hollow or solid brick with the size of (200 mm x 100 mm x 70 mm). To construct the wall, bricks are separated by plaster, 30 mm vertical and 50 mm horizontally, while indoor plaster of 30 mm thick and outdoor plaster of 20 mm thick is applied as shown in the following figure. The hollow brick contains an air-pocket of 160 mm x 70 mm x 50 mm at the center, where solid brick has no air-pocket. Consider the outdoor temperature of 33 °C and 23 °C at indoor. The temperatures of internal and external wall are 25 °C and 29 °C respectively. Temperature difference between the two surfaces of the air gap inside the brick is 2 °C. The Nusselt number for convection heat transfer inside the air-pocket can be evaluated as, $Nu_L = 0.22[Pr(Ra_L)/(0.2 + Pr)]^{0.28}(H/L)^{-0.25}$. Neglect the radiation heat transfer from the internal and external surfaces of the wall as well as the internal surfaces of the air gap inside the bricks.

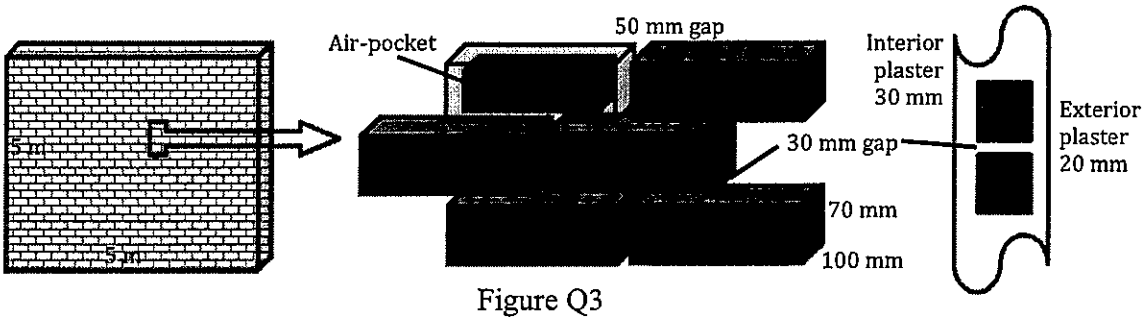


Figure Q3

Assume the air density, kinematic viscosity, thermal conductivity, thermal expansivity (isobaric volume expansivity), and specific heat are 1.23 kg/m^3, 1.46 x 10^{-5} m^2/s, 0.0246 W/m·K, 0.00333 K^{-1}, and 1.005 kJ/kg·K respectively. Stefan–Boltzmann constant and gravitational acceleration are 5.67 x 10^{-8} W/m^2·K^4 and 9.81 m/s^2, respectively. The thermal conductivity of the brick, plaster, and wood fiberboard are 0.72 W/m·K, 0.22 W/m·K, and 0.057 W/m·K respectively.

a. Find the thermal resistance and the rate of heat transfer through the wood fiberboard wall.

[3 marks]

b. Determine the thermal resistance and rate of heat transfer by conduction for the wall using hollow bricks.

[15 marks]

c. Determine the thermal resistance and rate of heat transfer by conduction for the wall using solid bricks.

[5 marks]

d. Calculate the percentage of energy saving for air conditioning by replacing the wall using hollow and solid bricks.

[2 marks]

Continued...

Question 4

A 2 m x 2 m single layer glass with 10 mm thickness is used as roof windows in the warehouse. The roof window is installed horizontally on the concrete roof (16 m x 6 m) of the warehouse as shown in the following figure. At night, the indoor temperature of the warehouse is kept at 20 °C. The indoor glass temperature is 23 °C and the outdoor air temperature is 27 °C. The effective sky temperature for radiation exchange between the both glass windows and concrete roof is 0 °C. Ignore the radiation exchange between the ground, nearby building, as well as inside the warehouse.

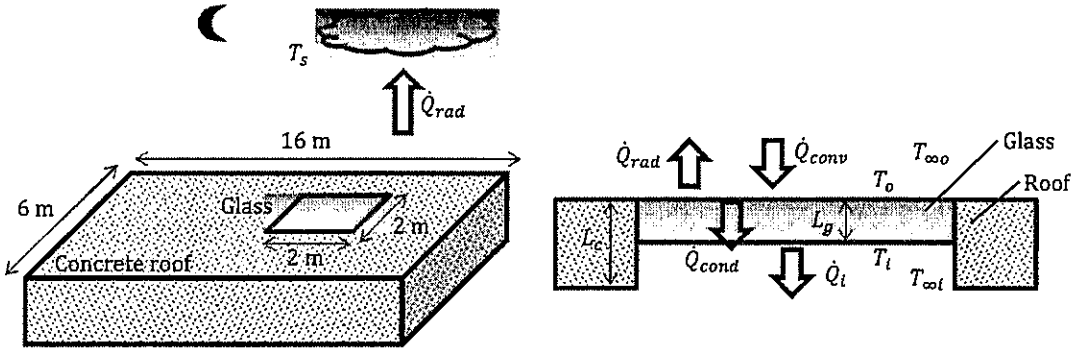


Figure Q4

Assume ideal gas for air where the air kinematic viscosity, thermal conductivity and thermal diffusivity are 1.6 x 10^{-5} m^2/s, 0.026 W/m·K and 2.3 x 10^{-5} m^2/s respectively. The thermal conductivity and emissivity of the glass are 0.78 W/m·°C and 0.85 respectively. The thermal conductivity and emissivity of the concrete are 0.82 W/m·°C and 0.75 respectively. Given the Stefan–Boltzmann constant and gravitational acceleration are 5.67 x 10^{-8} $W/m^2 \cdot K^4$ and 9.81 m/s^2 respectively.

a. Determine the rate of heat conduction through the glass.

[9 marks]

b. Calculate the heat loss through radiation of the glass.

[4 marks]

c. Find the heat transfer by outdoor convection of the glass and hence the Nusselt number.

[5 marks]

c. Calculate the rate of heat transfer by radiation and convection of the concrete roof, if the surface temperature of concrete roof at indoor is 23 °C.

[7 marks]

Continued...

APPENDIX A : SUMMARY OF FORMULAS

A1. Relation between relative humidity, absolute humidity, dry-bulb and wet-bulb temperature:

$$\phi = \frac{\omega}{\omega + 0.622}\left(\frac{p_T}{p_{v*}}\right) ; \omega = \frac{0.622\phi p_{v*}}{p_T - \phi p_{v*}} ; \omega = \frac{\left(\frac{0.622}{p_T/p_{v*@Twb} - 1}\right) h_{fg@Twb} - c_{pa}(T_{db} - T_{wb})}{h_{g@Tdb} - h_{f@Twb}}$$

A2. Friction factor for pipe flow in term of Reynolds number and roughness:

Laminar	Turbulent smooth, $\xi^+ = 0$	Turbulent minor roughness, $0 < \xi^+ < 4$	Turbulent moderate roughness, $4 \le \xi^+ \le 60$	Turbulent fully rough, $\xi^+ > 60$
$f = \frac{64}{Re_D}$	$f = 0.316(Re_D)^{-\frac{1}{4}}$	$\frac{1}{\sqrt{f}} = -1.80\log\left(\frac{6.9}{Re_D}\right)$	$\frac{1}{\sqrt{f}} = -1.8\log\left[\left(\frac{\xi^+}{3.7}\right)^{1.1} + \frac{6.9}{Re_D}\right]$	$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\xi^+}{3.7}\right)$
Laminar, $Re_D < 2000$ Turbulent, $Re_D \gg 2000$		$\frac{1}{\sqrt{f}} = -2.0\log\left[\frac{2.51}{(\sqrt{f})(Re_D)}\right]$	$\frac{1}{\sqrt{f}} = -2.0\log\left[\frac{\xi^+}{3.7} + \frac{2.51}{(\sqrt{f})(Re_D)}\right]$	

A3. Reynolds number, Prandtl number, Nusselt number, Grashof, and Rayleigh number:

$$Re = \frac{UL}{\nu} ; Pr = \frac{\nu}{\alpha} = \frac{\rho\nu c_p}{k} ; Nu = \frac{hL}{k} ; Gr = \frac{g\beta(T_\infty - T_s)L^3}{\nu^2} ; Ra = Gr \cdot Pr$$

A4. Quasi-one-dimensional heat conduction equation:

$$\dot{Q}_x = -kA\frac{\partial T}{\partial x} ; \frac{\partial T}{\partial t} = \alpha\frac{\partial^2 T}{\partial x^2}$$

A5. Nusselt number of force convection over flat plate:

$$Nu_L = \begin{cases} 0.6641\, Pr^{\frac{1}{3}} Re_L^{\frac{1}{2}} & Re_L \le 5\times10^5 \\ Pr^{\frac{1}{3}}\left(0.037\, Re_L^{\frac{4}{5}} - 871.2\right) & Re_L \ge 5\times10^5 \\ 0.037\, Pr^{\frac{1}{3}} Re_L^{\frac{4}{5}} & Re_L \gg 5\times10^5 \end{cases}$$

A6. Nusselt number of free convection for vertical plate:

$$Nu_L = \begin{cases} \frac{4}{3}\left(\frac{Gr_L}{4}\right)^{\frac{1}{4}}\left[\frac{0.75\sqrt{Pr}}{(0.609 + 1.221\sqrt{Pr} + 1.238\, Pr)^{1/4}}\right] & Ra_L \le 10^9 \\ \left[0.825 + 0.387\, Ra_L^{\frac{1}{6}}\left[1 + (0.492/Pr)^{\frac{9}{16}}\right]^{-8/27}\right]^2 & Ra_L > 10^9 \end{cases}$$

A7. Nusselt number of free convection for horizontal plate:

Warm surface face up / Cold surface face down

$$Nu_L = \begin{cases} 0.54\, Ra_L^{\frac{1}{4}} & 10^4 \le Ra_L \le 10^7, Pr \ge 0.68 \\ 0.15\, Ra_L^{\frac{1}{3}} & 10^7 \le Ra_L \le 10^{11}, Pr \ge 0.68 \end{cases}$$

Cold surface face up / Warm surface face down

$$Nu_L = \begin{cases} 0.52\, Ra_L^{\frac{1}{5}} & 10^4 \le Ra_L \le 10^7, Pr \ge 0.68 \\ 0.27\, Ra_L^{\frac{1}{4}} & 10^7 \le Ra_L \le 10^{11}, Pr \ge 0.68 \end{cases}$$

A8. Nusselt number of free convection inside enclosure:

Horizontal enclosure with cold surface on top

$$Nu_L = \begin{cases} 0.195\, Ra_L^{\frac{1}{4}} & 10^4 \le Ra_L \le 4\times10^5 \\ 0.068\, Ra_L^{\frac{1}{3}} & 4\times10^5 \le Ra_L \le 10^7 \end{cases}$$

Horizontal enclosures with warm surface on top

$$Nu_L = 1$$

Vertical enclosure

$$Nu_L = \begin{cases} 0.18\left[\frac{Pr(Ra_L)}{0.2 + Pr}\right]^{0.29} & Ra_L \ge 10^3(1 + 0.2/Pr), 10^{-3} \le Pr \le 10^5, 1 \le H/L \le 2 \\ 0.22\left[\frac{Pr(Ra_L)}{0.2 + Pr}\right]^{0.28}(H/L)^{-\frac{1}{4}} & 10^3 \le Ra_L \le 10^{10}, Pr \le 10^5, 2 \le H/L \le 10 \\ 0.42(Ra_L)^{\frac{1}{4}} Pr^{0.012}(H/L)^{-0.3} & 10^4 \le Ra_L \le 10^7, 1 \le Pr \le 2\times10^4, 10 \le H/L \le 40 \end{cases}$$

A9. Radiation heat transfer of gray surface:

Large Cavity	Large Parallel Planes	Long Concentric Cylinders	Concentric Spheres
$\dot{Q}_{12} = \varepsilon_1\sigma A_1(T_1^4 - T_2^4)$	$\dot{Q}_{12} = \frac{\sigma A_1(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1}$	$\dot{Q}_{12} = \frac{\sigma A_1(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1-\varepsilon_2}{\varepsilon_2}\left(\frac{r_1}{r_2}\right)}$	$\dot{Q}_{12} = \frac{\sigma A_1(T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1-\varepsilon_2}{\varepsilon_2}\left(\frac{r_1}{r_2}\right)^2}$

*Other information can be obtained from the property tables.

End of Paper